Vietnam Journal of Mechanics, Vietnam Academy of Science and Technology
DOI: https://doi.org/10.15625/0866-7136/14904

TUNING ACOUSTIC PERFORMANCE OF MULTI-CHAMBER HYBRID MUFFLERS

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Received 18 March 2020 / Published online: 05 July 2021

Abstract. In this paper, we investigate the functional acoustic performance of multi-chamber mufflers using a numerical approach. The wave propagation governing in expansion chamber domains is first introduced and solved by the finite element method. Our numerical results of selected muffler configurations are compared with the reference predictions model and experiments in order to validate the present procedure. Then, the influence of the geometry characteristics of typical and hybrid configurations of multi-chambered mufflers (number of sub-chambers, micro-perforated tube structure) on their sound transmission loss is studied. The obtained results indicate that the structure of the considered muffler has a strong effect on their acoustical performance, and the location and the high level of resonances of the sound transmission loss behavior are strongly related to the number of sub-chambers as well as micro-perforated tube characteristics. By tuning geometrical parameters (e.g., having a small perforation ratio), we enable to design mufflers having a higher sound transmission loss (up to 110 dB) at low frequencies ($\sim$ 195 Hz) but a constraint space (e.g., acoustic chamber length of 300 mm).

Keywords: exhaust muffler, low frequency, sound transmission loss, partition, micro-perforated tube.

1. INTRODUCTION

Mufflers or silencer are the primary and potential solutions for noise reduction purposes in a variety of industrial fields (e.g., exhaust engines or industrial systems). Expansion chamber mufflers have been used extensively in real industrial applications due to their prime function of noise treatment [1, 2]. How these devices are to the best of their acoustical performance as well as other working functions (i.e., chemical and thermal properties [3]) is the research question addressed in many studies.

Different approaches have been developed in the literature to predict the link between the geometrical parameters of expansion mufflers and their acoustical performance: analytical, numerical, and experimental approaches. The analytical method (e.g., transfer matrix method, effective medium model) focuses on finding a theoretical solution, leading to a better understanding of the mathematical and physical bases of the macroscopic equations governing acoustic dissipation phenomena in a muffler domain with simple shapes (single inlet/outlet [4,5], or single-inlet/double-outlet [6]). It can be stated that these analytical models often make simplifications on the displacement field and geometry, thereby imposing limitations on the type of problems to be solved. The numerical method (based on the computational techniques such as finite element method (FEM) [7–9] and boundary element method (BEM) [10, 11] have also been successfully used to predict the acoustic attenuation performance of mufflers with a variety of geometrical characteristics. The third category characterizes the acoustical performance of actual mufflers by measuring their sound transmission loss. Two alternative measurement approaches are widely used [12] either the two-load method [13] or the two-source method [14].

As demonstrated in a number of reference studies, the acoustic behavior of chamber mufflers strongly depends on the geometrical aspects such as the chamber shape [4, 5, 15, 16], the area ratio between ducts and their relative position [2, 15, 17]. It can be also stated that the level of sound reduction after attaching the muffler is strongly related to other general geometrical factors (e.g., body/pipe diameter, length/pipe diameter), chamber configuration (e.g., shape and connection of acoustic chambers). In recent years, micro-perforated panel absorbers have attracted much attention as promising alternatives to traditional sound absorbing materials. Its acoustic impedance can be reasonably well predicted using an analytical model developed in Ref. [18], which regards the small perforation hole as a lattice of short narrow tubes, with an end correction term being added to account for the attached air mass on both ends. The attempt of using micro-perforated structures for noise control inside mufflers was made by a number of studies [19–22].

It can be stated that using either internal partition or micro-perforated tube solution can provide a great improvement in the acoustic performance of the mufflers. However, within a fixed space, individually and collectively performances of these design solutions remain a problem to be examined. In addition, some constraints in the manufacturing process (e.g., the diameter of perforated hole > 1 mm) and working conditions (e.g., keeping the straight flow path) should be considered to meet the design requirements of a muffler. Thus, the present work deals with a numerical analysis on the acoustic property (sound transmission loss) of multi-chamber hybrid mufflers.

In the present work, the numerical method is employed to solve the wave propagation problem in muffler acoustic chambers. Firstly, the physical model of wave traveling is introduced by the governing equation in the chamber domain, and the mathematical description of its acoustical performance (i.e., sound transmission loss) is provided. Secondly, in the validation step, the numerical results of circular single chamber muffler configurations are compared with those proposed in the reference works, showing a good agreement. Then, the influence of the geometry characteristics on the acoustic performance of some typical and hybrid multi-chamber mufflers is investigated. Finally, some concluding remarks are stated.

2. METHODOLOGY

2.1. Structure designs

Considering the sound reduction capacity after installing, the industrial and exhaust mufflers can be cataloged as [23]: residential (20 ÷ 30 dB), critical (25 ÷ 35 dB), and super critical grade (35 ÷ 45 dB). Dependence of the noise resources and working requirements, the mufflers are designed appropriately. In terms of the structural configuration, there are several kinds of mufflers: (i) with a single inlet/outlet or with more an inlet/outlet pipe; (ii) with a single chamber or multi-chamber structures; (iii) with and without absorbing structures (e.g., porous material, micro-perforated tube/wall) or dividing/tuning components (e.g., V-blade, resonant chamber, tube, wall). Noted that in real applications, some mufflers with hybrid geometry configurations (e.g., turbo structures with a circular flow path [3]) are used.

In this work, we restrict our consideration to the hybrid partition mufflers (Fig. 1). All proposed mufflers have the same chamber as a straight-circular shape, but the different number of partitions and micro-perforated tubes inside the main acoustic chamber. They are three types: the original single chamber muffler (Fig. 1(a)), the five-chamber muffler (Fig. 1(b)), the five-chamber muffler with micro-perforated tubes (MPTs) (Fig. 1(c)). Noted that the inlet and outlet pipes of these mufflers have the same length and diameter. The different interesting acoustic performance of these mufflers will be characterized in the following.

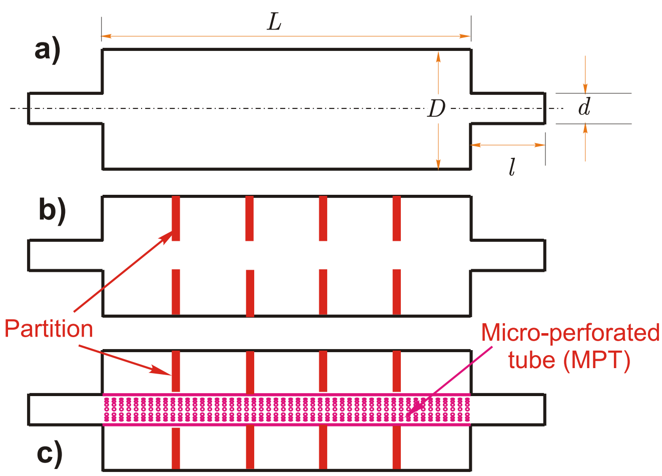

Fig. 1. Various geometrical configurations of multi-chamber expansion mufflers

2.2. Mathematical formulation and numerical modeling

The acoustic domain and boundary of single chamber expansion muffler are shown in Fig. 2(a), with a two-dimensional (2D) illustration. In which, Ω is the full domain filled with fluid, boundary Γ_1 and Γ_2 are respectively the inlet and outlet surface of the muffler, and $\partial\Omega$ is the fluid-solid interface that could be the muffler housing or the partition surfaces added inside. Due to the symmetry property of the muffler structure, the one-quarter model is used to reduce the computing cost (see symmetry boundary Γ_0 shown in Fig. 2(a)).

The steady-state problems in frequency domain, known as Helmholtz equation, is governed in the muffler acoustic chamber Ω [1,2]

$$\nabla^2 p + k^2 p = 0, \tag{1}$$

where p is the acoustic pressure, k is the wave number defined as the ratio of the angular frequency ω and the sound speed in the air c, and ∇^2 is the Laplacian operator.

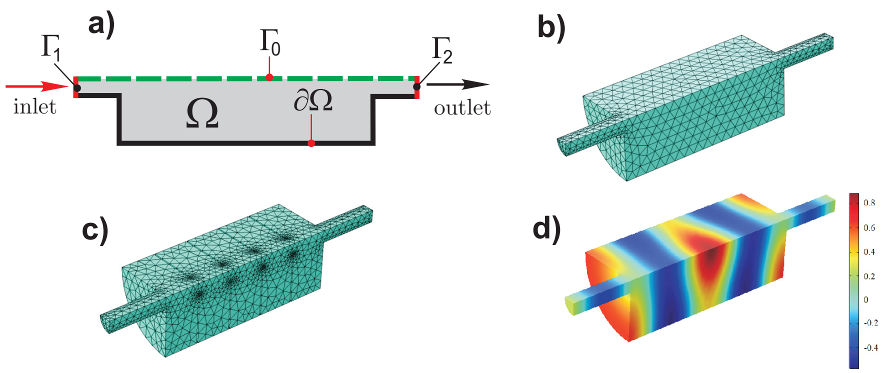

Fig. 2. (a) 2D illustration: acoustic domain and boundaries of a single chamber expansion muffler; (b) and (c) FE mesh model of single chamber and 5-chamber muffler including 9419 and 20796 tetrahedral elements, respectively; and (d) an illustrative solution of the acoustic pressure field

In order to solve Eq. (1), three following boundary conditions are implemented [24]. An incoming-outgoing plane waves is assumed at the inlet boundary Γ_1

$$\nabla p \cdot n = jkp - jkp_0, \tag{2}$$

where $j = \sqrt{-1}$ is the imaginary unit, n is the normal direction unit vector, and p_0 is the applied pressure at entrance to the inlet pipe.

At the downstream end of the finite element model Γ_2, the pipe section was anechoically terminated by applying the following condition

$$\nabla p \cdot n = jkp. \tag{3}$$

The hard-wall boundary condition are applied at muffler remaining surfaces $\partial\Omega$, separating walls between the resonating chambers (Fig. 1(b)), and the walls of the micro-perforated tube (Fig. 1(c)) added

$$\nabla p \cdot n = 0. \tag{4}$$

As a muffler characteristic property and independent on the internal flow conditions, sound transmission loss (TL) is usually referred to the accumulated decrease in intensity of waveform energy as a wave propagates outwards from a source, or as it propagates through a certain area or through a certain type of structure. Sound transmission loss is defined as the incident (W_{in}) sound power over the transmitted (W_{out}) sound power as [1]

$$\mathrm{TL} = 10\log\left(\frac{W_{in}}{W_{out}}\right), \tag{5}$$

in which these sound powers are estimated the acoustic pressure as

$$W_{in} = \iint_{\Gamma_1} \frac{p_0^2}{2\rho c} d\Gamma, \; W_{out} = \iint_{\Gamma_2} \frac{p_{tr}^2}{2\rho c} d\Gamma. \tag{6}$$

In this work, all finite element (FE) computations are performed using COMSOL Multiphysics. The above numerical framework can be proceeded as follows:

- For the geometrical feature, except for the validation cases (Section 2.3, some parameters (L, D, l, and d) of the considered mufflers are kept during the investigation step (see Fig. 1(a)). Several remaining factors (e.g., number of partitions, hole diameter and distribution) will be considered as tuned ones.

- The frequency range of interest is [20 2500] Hz, and the transmission loss performance will be characterized.

- Based on the muffler geometry, the governing equations over its fluid domain and boundary conditions (see Fig. 2(a)) are implemented. Then, the corresponding FE mesh model is generated (Fig. 2(b)–(c)), and the desired field solutions (i.e., sound pressure) can be obtained (see Fig. 2(d)). Thus, the sound transmission loss of mufflers is estimated by the integral operator.

- For each muffler configuration, the convergence analysis versus the number of finite elements is examined at several individual frequencies (see Section 2.4).

2.3. Validation and verification

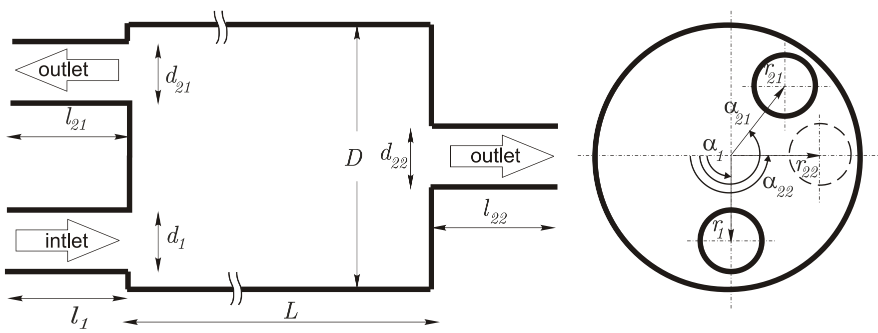

Fig. 3. Geometry parameters of a single inlet/double outlet muffler

In order to validate the present numerical approach, a reference geometry of single inlet/double outlet muffler with its analytical model (see Eq. (21) in Ref. [17], and Eq. (19) in Ref. [8]) is first considered. Several geometrical parameters of this reference muffler are: $L = 300$ mm; $l_1 = l_{21} = l_{22} = 100$ mm; $D = 183.75$ mm; $d_1 = d_{21} = d_{22} = 40$ mm; $r_1 = r_{21} = r_{22} = 45$ mm; $\alpha_1 = \pi$, and $\alpha_{21} = \alpha_{22} = 0$ (see Fig. 3). As the comparison presented in Fig. 4(a), the FE calculations and reference analytical model show an excellent agreement. Next, we consider a single inlet/outlet muffler (see, Fig. 1(a), geometry factors are $D = 6.035$ inch, $d = 1.375$ inch, $L = 8$ inch, and $l = 1.5$ inch). Our numerical computation is compared with the available analytical model (see Eq. (4.26) in Ref. [1]), numerical (BEM) and experimental results of transmission loss [12]. It can be seen from Fig. 4(b) that our FE prediction is very close with both reference experimental data and numerical estimation. The discrepancy between the analytical model and other curves in range frequency large than 1800 Hz (see the thin line in Fig. 4(b)) shows that the effect of higher order modes on the transmission loss could be not negligible. These reference examples show a strong validation for the proposed finite element procedure as well as its potential implementation.

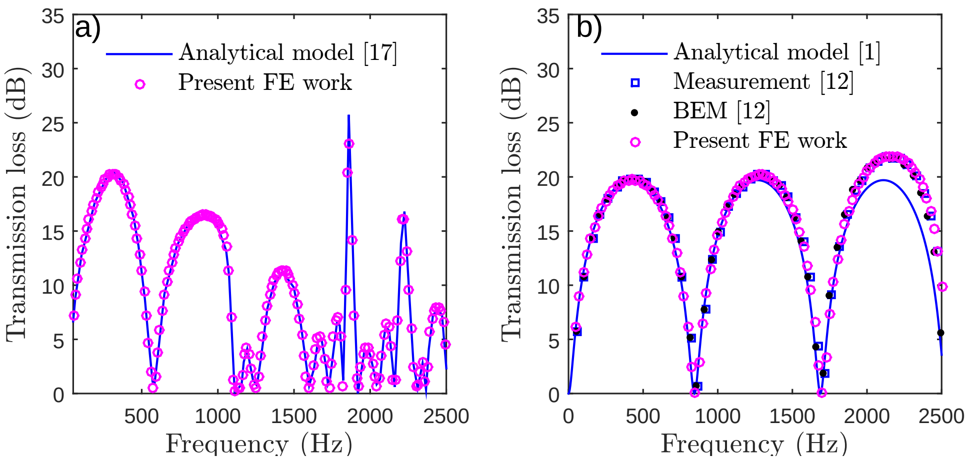

Fig. 4. The present FE predictions (circle markers) of the sound transmission loss compared with reference results for mufflers with (a) single and (b) double outlet

2.4. Convergence analysis

In this section, a convergence analysis for FE computations of permeability is present. We introduce the relation between the difference of the computed transmission loss, $d\text{TL}$, and the difference of the corresponding FE numbers of element, dN_e, at mesh levels $(i+1)$ and (i). From the results, we obtain the same following trends for all considered cases

$$\frac{d\text{TL}}{dN_e}\left(=\frac{\text{TL}^{(i)}-\text{TL}^{(i+1)}}{N_e^{(i+1)}-N_e^{(i)}}\right)=e^a N_e^b, \tag{7}$$

where a and b are the fitting coefficients given. From this, the error of FE computation at a mesh level N_e can be estimated as

$$\epsilon_{fem}\left(=\frac{\text{TL}^{(N_e)}-\text{TL}^{(N_{e,\infty})}}{\text{TL}^{(N_e)}}\right)=\frac{1}{\text{TL}^{(N_e)}}\int_{N_e}^{N_{e,\infty}} e^a N_e^b dN_e. \tag{8}$$

The error ϵ_{fem} in Eq. (8) at mesh level of $N_{e,\max}$ can be estimated from

$$\epsilon_{fem} = \frac{e^a}{\mathrm{TL}^{(N_{e,\max})}} \frac{N_{e,\infty}^{b+1} - N_{e,\max}^{b+1}}{(b+1)}, \tag{9}$$

within $N_{e,\infty} \to \infty$ and $b < -1$, we obtain

$$\epsilon_{fem} = -\frac{e^a}{\mathrm{TL}^{(N_{e,\max})}} \frac{N_{e,\max}^{b+1}}{(b+1)}. \tag{10}$$

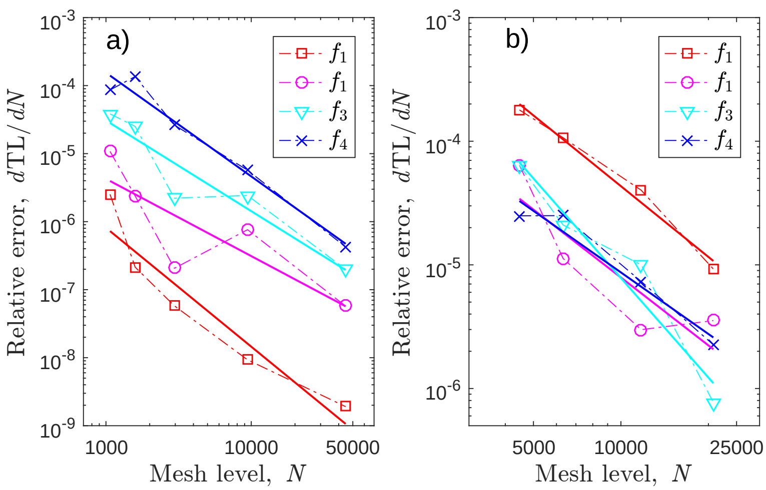

Fig. 5. Results of convergence analysis in TL calculations at several frequencies for (a) single chamber and (b) five-chamber muffler

Table 1. Convergence characteristics of FE computations of the transmission loss

Muffler	f (Hz)	a	b	R^2	$\mathrm{TL}^{(N_{e,\max})}$ (dB)	$N_{e,\max}$	ϵ_{fem}
1-Sub	230	−1.948	−1.748	0.9023	21.55	44611	1.4e−5
	830	−4.519	−1.136	0.7059	22.14		2.3e−3
	1430	−1.183	−1.333	0.8962	23.11		2.4e−3
	1880	1.761	−1.525	0.9723	21.93		3.1e−3
5-Sub	775	7.461	−1.901	0.9826	97.99	20796	5.8e−2
	1280	5.051	−1.824	0.7711	152.29		6.9e−3
	1505	12.880	−2.675	0.9390	113.48		1.4e−3
	1885	3.580	−1.654	0.9512	62.53		7.4e−3

Herein, we illustrate two muffler configurations with the FE mesh models shown in Fig. 2(b)–(c). The obtained convergence characteristics are provided in Fig. 5, in which the results of convergence analysis in calculations of the transmission loss at a set of four selected frequencies around the resonance locations of the TL behavior. Using Eq. (10), we can estimate the relative error for a given mesh level (see, Table 1). For all testing cases, the tolerance error $\epsilon_{fem} < 10^{-2}$ is estimated.

3. RESULTS AND DISCUSSION

In this section, sound transmission loss of several chambered muffler without and with micro-perforated tubes are investigated. Structure of muffler has fixed dimensions as: the total acoustic chamber with a length of $L = 300$ mm and a diameter of $D = 2L/3$, both inlet and outlet pipes within a length of $l = L/3$ and a diameter of $d = 2L/15$. First, we investigate the effect of number of internal partitions or sub-chambers on the muffler acoustical performance. Then, MPTs are adapted into the 5-chamber muffler to exam the effects of micro-perforated configurations on the transmission loss behavior.

3.1. Typical multi-chamber mufflers

Fig. 6 compares the sound transmission loss of three multi-chamber mufflers (see Fig. 1(b) for a case having five sub-chambers) with that from the single chamber one (Fig. 1(a)). It can be seen that by adding partitions inside chamber expansion the noise reduction level of mufflers is improved significantly. The TL performance increases from value of 20 dB (circle markers) up to than 40 dB, 75 dB, 189 dB, and 230 dB corresponding the mufflers within 1, 2, 3, and 4 internal partitions. It is also noted that the number of TL resonances within sharp peaks decreases within an increase of the used partitions. That leads to the broaden of high TL over a large frequency band in muffler with higher sub-chambers (e.g., 5-chamber muffler with TL than 45 dB over frequency band ranging from 560 Hz to 1970 Hz, see the thickest continuous line in Fig. 6). Interestingly, the higher partitions used the higher and the broader the TL sharp peak obtained. From the obtained results, it can be also noted that muffler having the fixed geometry parameters

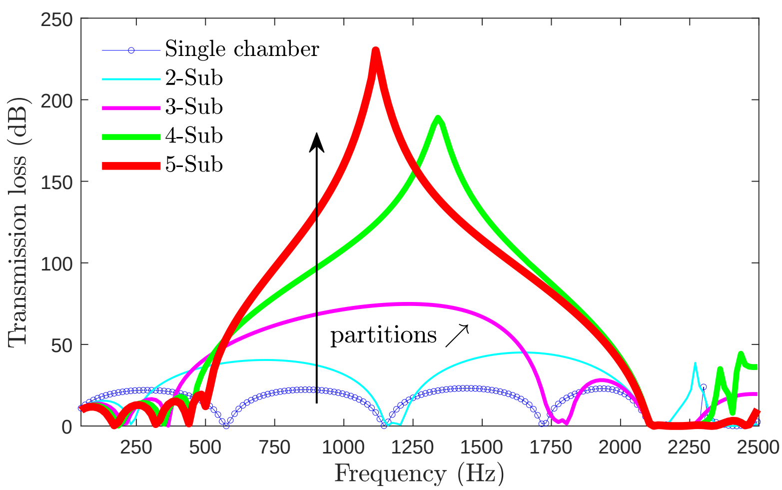

Fig. 6. Influence of the sub-chamber number on the muffler sound transmission loss

we can only tune their TL performance at discrete frequency range due to the integer of the partition or sub-chamber number, and the poor sound transmission loss at low frequencies could not be improved by adding partitions. These limitations can be solved in the below.

3.2. Multi-chamber mufflers using micro-perforated tubes

The improvement of the acoustical performance of mufflers having micro-perforated tube is examined by adding MPTs within multi-chambers (see, Fig. 1(c)). Noted that these sub-chambers are covered by the MPT with geometry configurations of the hole diameter d_h and the perforation area ratio ϕ_p defined as

$$\phi_p = \frac{d_h^2 N_d N_l}{4Ld}, \tag{11}$$

in which N_d and N_l are numbers of micro-perforated holes distributed radially and axially in tubes. The value $N_d = 12$ is selected for all muffler configurations, whereas the number N_l is a tuned parameter.

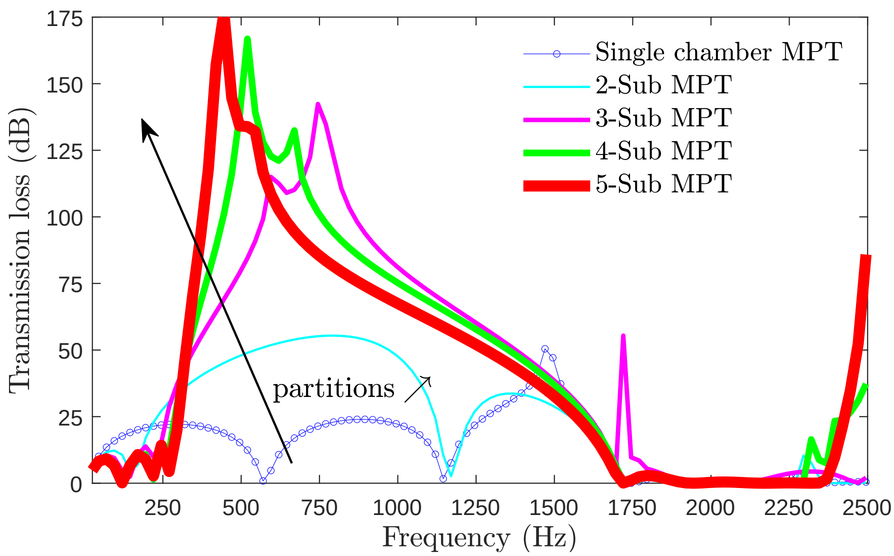

Fig. 7. Influence of number of partitions and micro-perforated tube on the muffler TL performance

Fig. 7 shows the influence of the MPT (having $d_h = L/100$ (i.e., $d_h = 3$ mm), $N_l = 16$ (corresponding $\phi_p = 3.6\%$) on the TL performance of various multi-chamber mufflers. Compared with the TL performance in Fig. 6, we can see clearly that using micro-perforated tube we can design mufflers for low frequency applications (e.g., ~ 350 Hz). The investigated curves show that the five sub-chamber muffler has the better sound transmission loss in compared with the remaining ones. Having the considered MPT, the slight difference between mufflers having four and five sub-chambers indicate that there is a TL limitation (i.e., within a TL peak around frequency of 500 Hz) for these mufflers.

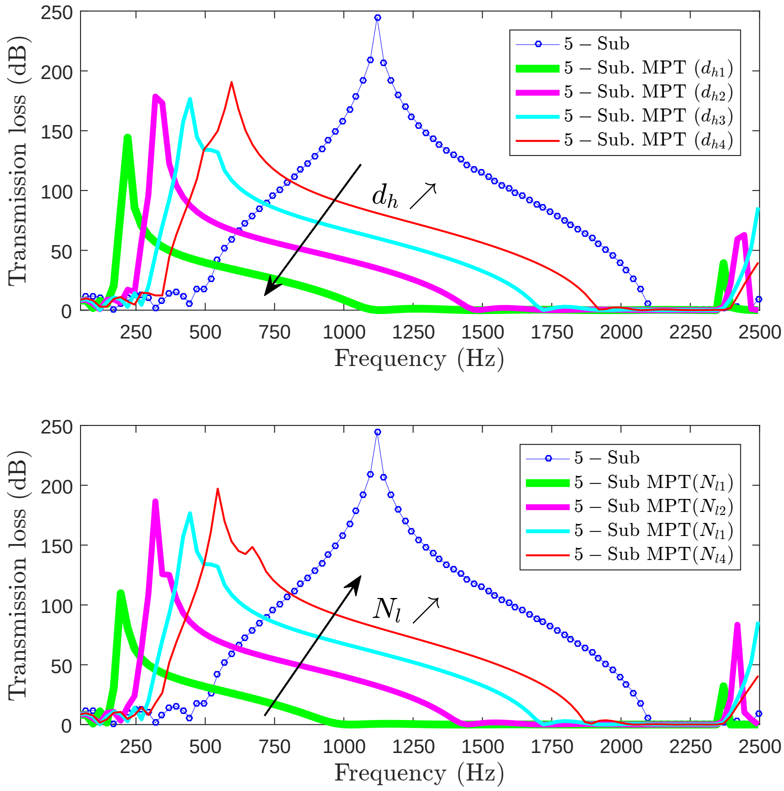

Fig. 8. Influence of (top) the hole diameter d_h and (bottom) the axial hole number N_l on the sound transmission loss of five sub-chamber mufflers

The influence of the MPT characteristics on TL performance of five sub-chamber mufflers is next studied. The top part of Fig. 8 is for varying the diameter of micro-perforated holes (named the MPT1 muffler class with $d_h = [1,\ 2,\ 3,\ 4]$ mm and $N_l = 16$ so the corresponding perforation ratio $\phi_p = [0.4,\ 1.6,\ 3.6,\ 6.4]$ %), whereas the bottom is for their axial number of hole (named the MPT2 muffler class with $N_l = [6,\ 10,\ 16,\ 25]$ and $d_p = 3$ mm corresponding $\phi_p = [1.4,\ 2.3,\ 3.6,\ 5.6]$%). In general, we can see that the lower perforation ratio added (due to either micro-perforated hole diameter or number of holes) the lower frequency occurring and the narrower the sharp peak of TL property archived. In detailed, we can overcome the low frequency limitation of 500 Hz above-described by a lower value of 200 Hz, at this frequency the TL capacity of designed mufflers can reach a value of $\sim$ 145 dB at 220 Hz (for MPT1 with $d_p = 1$ mm and $N_l = 16$, top part of Fig. 8) and $\sim$ 110 dB at 195 Hz (for MPT2 with $d_p = 3$ mm and $N_l = 6$, bottom part of Fig. 8). In these two better mufflers, the second muffler with a higher $\phi_p = 1.4\%$ is more convenient and reasonable in the manufacturing as well as working conditions in compared with the first one within $\phi_p = 0.4\%$.

4. CONCLUSIONS

The present work suggested and developed a thee-dimensional finite element model for predicting the acoustical performance of hybrid multi-chamber mufflers with and without micro-perforated elements. Overall, the design of advanced mufflers shows a better sound transmission loss performance in compared with the behavior of the single chamber structure.

From the preceding analysis, the following statements can be drawn. The lost energy of sound wave propagating inside mufflers is related to the characteristics of the flow path inside. Both inertial partition and micro-perforated tube configurations can provide a better TL over the considered frequency range. As partitioning a muffler main chamber into sub-chambers, the individual resonances of its original TL performance seem to be merged. That leads to the TL sharp peak on multi-chamber muffler.

For real applications, the present study reveals that five sub-chamber mufflers with a very small perforation ratio can provide a high sound transmission loss capacity for noise at low frequency.

ACKNOWLEDGMENT

This work is funded by the Vietnam National Foundation for Science and Technology Development (NAFOSTED) under grant number 107.01-2019.316.

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